ROBUST APPROACH FOR SHELL AND TUBE EXCHANGERS OPTIMIZATION UNDER UNCERTAIN HEAT TRANSFER ESTIMATION

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Abstract. *In this paper a procedure for optimal design of shell and tube heat exchangers considering uncertainties in the estimation of heat transfer coefficients is proposed. The methodology adopts a genetic algorithm to determine the best equipment architecture and design parameters in order to design a robust exchanger, i.e. to minimize the sensitivity of equipment performance to variations in the actual value of the overall heat transfer coefficient, or to optimize a user-defined objective function including capital investment, operating costs, value of the transferred heat, and penalties for unmet specifications, subject to random variations of the actual heat transfer coefficient according to predetermined probability density functions. The performance improvement obtained when passing from deterministic to uncertain design environment are then demonstrated resorting to some numerical examples.*

Keywords*: Heat exchanger design, genetic algorithm, optimization, uncertainty.*

1. INTRODUCTION

Heat exchangers are widely employed in industrial processes and their cost minimization, or the maximization of thermal performances, is an important target for both designers and users. Traditional design approaches are based on iterative procedures which gradually change design parameters until a satisfying solution which meets the design specifications is reached. However, such methods, besides being time consuming, do not guarantee the reach of an economically optimal solution. In recent times a renewed interest in the optimal design of heat exchangers has been thus witnessed in the literature. This corresponds to the availability of new optimization techniques, such as genetic algorithms, able to handle a large number of design parameters including both discrete and continuous variables (Babu and Munawar, 2007; Caputo et al., 2008a; Hilbert et al., 2006; Ponce-Ortega et al., 2009; Tayal et al., 1999). Nevertheless, a number of issues still remain to be solved. For instance, most computer aided optimal design procedures assume steady state operations thus neglecting either deterministic and stochastic variability in the operating conditions. The uncertainty in the heat transfer coefficients correlations is also neglected, and penalties for off-design performances are usually not included in the evaluation of economic objective functions. Finally, the cost correlations used for economic optimization are not sensitive to the constructive details of the equipment as determined by the design algorithm, but only to the overall heat exchange area. This justifies why some scholars are skeptical about the use of precise optimization methods when applied to heat exchanger design, owing to the inherent fuzziness of the problem given the uncertainty in operating conditions and in the adopted design correlations (Bell, 2000). Nevertheless, computer based optimization of heat exchangers is an established and widely accepted research subject, and the problem of optimal sizing of heat exchangers working under stochastic operating conditions or designed under uncertain heat transfer conditions is relevant in the wider field of optimization under uncertainty of process equipment (Sahinidis, 2004).

According to Polley and Pugh (2001) there have been two traditional approaches to dealing with heat transfer or design conditions uncertainties: over-specification of fouling resistance and addition of "design margin" (i.e. addition of extra surface area). This is equivalent to design for worst case conditions, but it may have strong drawbacks under the financial and operational point of view. As an alternative to over-design they suggest to size the equipment according to nominal heat transfer conditions, or a nominal "design point" if the equipment shall undergo variable process conditions, and then evaluate the responses of the exchanger to changed conditions. The approach is based upon recognition that although process plants are usually designed on the basis of operating at a 'point condition', they usually operate over a range of conditions. In case responses fall outside of an acceptable range a new design configuration is sought by changing the "design point" or the use of operational options (such as exchanger bypasses) are explored. Haseler et al. (1983) as well as Clarke et al. (2001) investigated the sensitivity of overall heat exchanger calculations to uncertainties in individual fluid properties, but they did not consider the effects of uncertainties in heat exchanger geometry or in process specifications, and did not addressed any optimization problem. James et al. (1995) applied uncertainty analysis to the prediction of cross-flow heat exchanger performances.

Taylor et al. (1999) used uncertainty analysis to determine bounds on the predicted performance parameters in thermal systems, while Bernardo et al. (2001) discussed the incorporation of robustness criteria in process equipment design problems under uncertainty. Affan Badar et al. (1993) used Monte Carlo analysis to assess the impact of design parameters uncertainties in the design of heat exchangers. They found that the overall heat transfer coefficient often is characterized by a Weibull distribution. Cho (1987), instead, presented a statistical-based method for sizing a heat exchanger based on the probability or confidence level that it will meet its intended thermal-hydraulic duty but, again, without any optimization approach. Knetsch and Hauptmanns (2005) analysed the effect of uncertainties in the dynamic response of heat exchangers demonstrating that the inclusion of stochastic effects and uncertainties provides a more reliable basis for design decisions. Finally, Shilling et al. (2009) propose a risk assessment method to evaluate the consequence of uncertain design and operational conditions. In order to contribute to a solution to the problem of designing heat exchanger subject to variable operating conditions Caputo et al. (2010) developed a genetic algorithm based procedure for design of shell and tube heat exchangers working under stochastic operating conditions. The method optimizes a user-defined objective function including capital investment, operating costs, value of the transferred heat, and penalties for unmet specifications. The objective function is computed factoring in, for any specific design configuration, the actual equipment performances obtained during off design operations caused by stochastic variations of process parameters according to predetermined probability density functions. In this paper, instead, a similar approach is adopted to design shell and tube heat exchangers in case of variation of the shell-side and tube-side heat transfer coefficients owing to uncertainties in the physical fluid properties or in the heat transfer correlations, which determine an uncertainty in the overall heat transfer coefficient and unexpected variations in the transferred heat. After describing the proposed design optimization method an application example is presented in order to verify the capability of the approach, showing that significant benefits are obtained when an exchanger optimized including design uncertainties is compared to a similar exchanger optimized assuming a constant and known value of the heat transfer coefficient.

2. METHODOLOGICAL APPROACH

In a previous work (Caputo et al. 2008a) a detailed computer model was developed for optimal design of shell and tube heat exchangers operating in stationary conditions and without uncertainties in heat transfer estimation, resorting to an optimization procedure based on genetic algorithm. That work was extended to take into account constructive details in the capital cost estimation (Caputo et al. 2008b, 2009), and to allow a joint optimization of exchanger design and cleaning schedule (Caputo et al. 2011). Finally, the model was upgraded to allow optimal design under deterministically or stochastically variable operating conditions (Caputo et al. 2010). The reader may refer to the above papers for details on the genetic algorithm implementation and the exchanger sizing procedure. In this paper we build on the approach used in previous works and extend our model to take into account uncertainties in the estimation of the heat transfer coefficients utilized for equipment sizing.

The computational procedure for optimal design under uncertainty includes the following steps.

- Input of design specifications and design duty.
- Generation of the values of a set of independent design variables (VIP) by the genetic algorithm.
- Computation of the nominal values of shell-side, h_s, tube-side, h_t, and overall, U, heat transfer coefficient.
- Computation of all remaining dependent design variables (VDP) and exchanger heat transfer area based on the required duty and other design specification.
- Generation of a probability density function for h_s and h_t in a given uncertainty range according to a predefined parametric probability distribution shape.
- Computation of the resulting probability distribution function of the overall heat transfer coefficient U.
- Evaluation of the capital investment, operating cost, and objective function by changing the actual U value according to the previously obtained probability distribution for the previously designed exchanger.
- Utilization of the optimisation algorithm to select a new set of values for the VIP.
- Iteration of the previous steps until a minimum of the objective function is found.

The entire process is schematised in Fig. 1. Design specification indicate the heat duty of the exchanger, and are given by imposing five of the following six parameters: the mass flow rates of the hot and cold fluids (m_c, m_f), as well as the inlet and outlet temperatures of the hot fluid t_{ci}, t_{cu}, and cold fluid, t_{fi}, t_{fu}, the remaining parameter being determined by an energy balance. All of the above process parameters are assumed to remain constant during the exchanger operation. Fixed parameters assigned by the user are the tubesheet patterns (triangular or square), the fouling resistances $R_{foul,\,shell}$ and $R_{foul,\,tube}$, and the assumed nominal thermophysical properties of both fluids.

Starting from this input data a random starting value is given to a set of independent design variables (VIP). In this model the Kern design method (Sinnott, 2005) has been implemented and the selected VIP are the tubes bundle diameter D_b, the tube outside diameter d_o, the central baffle spacing L_{bc}, the pitch ratio $L_{pt\,Ratio}$, the baffle cut B_c, the tube layout angle θ and the tube pass number N_{tp}.

The remaining heat exchanger's design features, i.e. the dependent design variables (VDP), are then directly computed from the VIP according to the selected design procedure. In particular, the shell-side and tube-side heat exchange coefficients h_s, h_t, the overall heat transfer coefficient including fouling resistance (U_{dirt}), the overall heat exchange area S, the number of tubes N_{tt}, the shell and tubes length L_{tt} and tube-side and shell-side flow velocities v_t and v_s, are determined, thus defining all constructive details of the exchanger satisfying the assigned thermal duty. The computed values of flow velocities and the constructive details of the exchanger structure are then used to evaluate the tube-side and shell-side pressure loss (DP_t and DP_s) which determine the operating costs, and the capital investment.

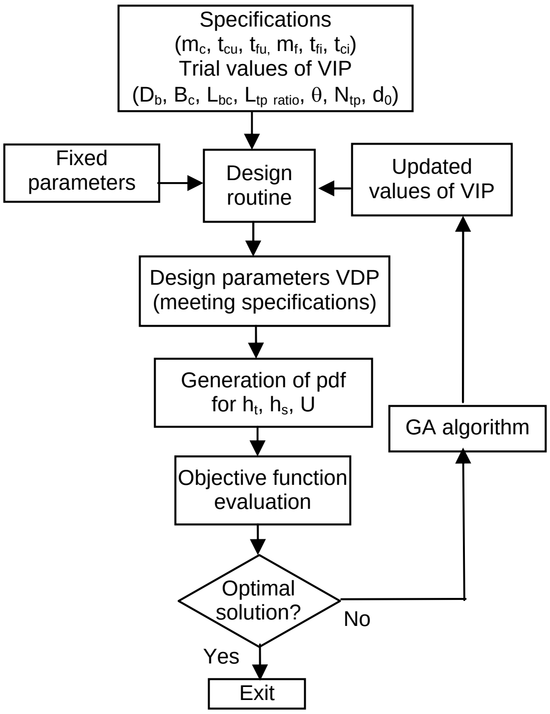

Figure 1. Optimized design procedure in case of uncertain heat transfer correlations.

Starting from the nominal computed values h_{ns} and h_{nt} of h_s, and h_t, their probability distribution function (pdf) is generated assuming a user-defined variation range around the nominal value and a shape of the pdf. Any shape of pdf which better suits the specific uncertainty environment can be used, but in this work it is assumed that h has a uniform distribution centred on its nominal value h_n, with maximum and minimum values defined as $h_{max} = (1+\alpha)\ h_n$ and $h_{min} = (1-\alpha)\ h_n$ respectively, where α is the maximum hypothesized percent variation, so that the entire variability range has an amplitude $2\alpha h_n$. The overall heat transfer coefficient values are computed by assigning values of h_s and h_t from their respective pdf, in Eq. (1),

$$U = \frac{1}{\frac{1}{h_s} + R_{foul,shell} + \frac{d_o}{d_i} \cdot \left(R_{foul,tube} + \frac{1}{h_t} \right)} \tag{1}$$

By generating all combinations of h_s, and h_t values within their variability ranges, all possible values of U are computed and the probability of each U value is obtained by multiplying the corresponding probability values of h_s, and h_t, given that the shell-side and tube-side heat exchange coefficients are considered as independent random variables. As a consequence the pdf of U is obtained with values ranging from U_{min} to U_{max}.

Given that the actual value of U will be different from the nominal value assumed in the exchanger design procedure, the equipment will behave in a different manner and design specifications could result exceeded or even unmet, and this has an economic consequence in terms of increased revenues or cost penalties.

According to each *i-th* value of the overall heat transfer coefficient U_i the corresponding value of the off-design economic performance $EP(U_i)$, which accounts for economic value of the benefits or penalties coming from exceeding or failing to meet some of the specifications during the actual off design operation, is then computed given the previously defined geometrical configuration of the heat exchanger. In this work the efficiency method is used to determine the actual condition of the output streams given the actual value of the heat transfer coefficient. Generally speaking the overall expected behaviour of the equipment, considering the uncertain variation of the heat transfer coefficient occurring in practice, can then be expressed by computing the expected value of the EP as follows assuming that distributions of h_s, h_t and U have been discretized in a finite number of values,

$$EP = \sum_{i=1}^{N} p_i EP(U_i) \qquad (2)$$

where N is the number of different U values included in its discrete pdf, $EP(U_i)$ is the value of the discounted economic performance corresponding to value U_i of the heat transfer coefficient and p_i is the probability that the *i-th* value of U occurs. This allows to compute the economic objective function (OF).

Given that the GA is designed to minimize an objective function representing the life cycle cost, this function is formulated in a different manner according to the case that the EP has a positive or negative value. In case, in fact, that EP determines a revenue instead of a cost penalty, then a maximization of the net equipment worth is to be sought instead of the minimization of the life cycle cost.

$$EP \le 0 \quad OF = \delta_{CI}\, C_I + \delta_{Ces}\, C_{es} + C_{fitt} - \delta_{EP}\, EP \qquad (3)$$

$$EP > 0 \quad OF = \frac{\delta_{CI}\, C_I + \delta_{Ces}\, C_{es} + C_{fitt}}{\delta_{EP}\, EP} \qquad (4)$$

where C_I is the capital investment, C_{es} is the is the total discounted present value of energy expenses for overcoming pressure drop caused by friction losses, C_{fitt} is a fictitious very high fixed cost penalty charged in case any of the design constraints on the allowed values of VDP is violated (this allows to quickly reject solutions not meeting some technical constraints), and δ_{CI}, δ_{Ces}, δ_{EP}, are coefficient set at the value 0 or 1 allowing the user to include or omit any of the cost items. The capital investment depends from the heat exchange area S (m^2) and equipment configuration. It can be estimated according to simplified correlations (Taal et al. 2003) such as Hall's equation

$$C_I(€) = 8000 + 259.2\, S^{0.91} \qquad (5)$$

However, when a detailed cost optimization is sought, more precise cost estimation techniques are required (Caputo et al. 2008b, 2009). The total discounted operating cost related to pumping power to overcome friction losses is

$$C_{es} = \sum_{k=1}^{ny} \frac{C_o}{(1+s)^k} \qquad (6)$$

$$C_o = P \cdot C_E \cdot N_{hr} \qquad (7)$$

where C_E [€/kWh] is the electric energy cost, N_{hr} [h/yr] the annual operating hours, P [W] the pumping power, *s* the annual interest rate [%/yr], ny [yr] the equipment life. P depends from the equipment pressure drop and, in turn, from the selected exchanger geometry, see Caputo et al. (2008a) for details. There are many ways to compute the EP according to the designer goals and the intended scope of the heat exchanger (be it for cooling or heating purposes, for heat recovery purposes, to maintain a given set point temperature for a stream etc.). In this work we assume that the heat exchanger duty is to cool a hot stream, and the following option for expressing on a yearly basis the economic performance has been adopted

$$EP_Y = \sum_{i=1}^{N} ES_i\, C_{kWhTF} \qquad (8)$$

where ES_i [kWh/yr] is the yearly amount of penalty or premium energy associated to the case that the *i-th* value of the heat transfer coefficient occurs, and C_{kWhTF} [€/kWh] is the specific cooling energy cost. Once EP_Y has been computed the discounted life cycle value of EP can be computed resorting to Eq. (6) by substituting EP_Y to C_O. The manner ES_i is computed depends on the user needs. Here the following sample options have been considered.

Case I) The exchanger is used to cool a stream in a "the more the better" manner and an economic revenue is associated to any degree of cooling obtained

$$ES_i = \left(t_{cu_R} - t_{ci}\right) m_c\, C_{pC}\, \left(N_{hr}\; p_i\right) \qquad (9)$$

where t_{cuR} [°C] is the actual outlet temperature of the hot stream corresponding to the value U_i of the heat transfer coefficient, t_{ci} [°C] is the inlet temperature of hot stream, m_c [kg/s] is the mass flow rate of hot stream, C_{pC} [kJ/kg K] is the specific heat of hot stream, N_{hr} [h/yr] is the number of yearly operating hours and p_i is the probability of occurrence of heat transfer coefficient value U_i.

Case II) The exchanger is used to cool a stream to a predefined target temperature t_{cuT} which can represent the design specification. If this temperature is not met then an economic penalty is applied equal to the additional cost borne to lower the actual fluid outlet temperature $t_{cuR}>t_{cuT}$ to the target temperature. If $t_{cuR} \leq t_{cuT}$ no penalty or premium is applied and $ES_i = 0$.

$$ES_i = \left(t_{cu_T} - t_{cu_R}\right) m_c\, C_{pC} \left(N_{hr}\; p_i\right) \tag{10}$$

Case III) In this case an economic premium is assigned when the hot stream is cooled below the target temperature t_{cuT}, but no penalty is applied if $t_{cuR}>t_{cuT}$ (i.e. in that case $ES_i = 0$). Equation (10) is used to compute ES_i.

Case IV) In this case an economic premium is assigned when the hot stream is cooled below the target temperature t_{cuT} and a penalty is applied if $t_{cuR}>t_{cuT}$. Equation (10) is used to compute ES_i.

When the goal of the designer is not to optimize the overall economic performance of the equipment, but rather to design a “robust” exchanger, i.e. one that is scarcely affected in its overall performances by the uncertainty of U values, or, to say it with different words, one where changes in h_s and h_t determine small deviations of U respect its nominal value U_n, then a minimization of $(U_{max}-U_{min})/U_n$ is sought and a design penalty in the objective function can be introduced as a function of the deviation of the value of heat transfer coefficient from its nominal value. In this manner the GA would seek a configuration with low variability of U in order to minimize the objective function. However, this case is not included in this work.

After computing the OF, the optimisation algorithm updates the trial values of the independent optimisation variables (VIP) which are then passed to the design routine to define a new architecture of the heat exchanger. The process is iterated until a minimum of the objective function is found or a prescribed convergence criterion is met as shown on the flow chart of Fig. 1. More details of the design procedure and the optimisation algorithm are given elsewhere (Caputo et al. 2008a,b).

3. APPLICATION EXAMPLE

To show the capability of the proposed method a case study is presented referring to an exchanger having design specifications shown in Table 1. The heat exchanger is of split ring floating head type (SRFH according to TEMA classification). Pumps efficiency is 0.8, N_{hr} = 7000 h/yr, C_E = 0.12 €/kWh, ny = 10 years, s = 10%/year, and C_{kWhTF} = 0.07 [€/kWh]. We assume that shell-side and tube-side heat transfer coefficients have a uniform distribution in a range ±30% respect their nominal value (α = 0.3).

Table 1. Design specification.

Hot stream inlet temperature t_{ci} [°C]	95.00	Cold stream inlet temperature t_{fi} [°C]	25.00
Hot stream outlet temperature t_{cu} [°C]	40.00	Cold stream outlet temperature t_{fu} [°C]	40.00
Hot stream mass flow rate m_c [kg/s]	27.77	Cold stream mass flow rate m_f [kg/s]	68.90
Threshold target cooling temperature t_{cuT} [°C]	42.00	LMTD correction factor Ft	0.81
Heat duty [kW]	4340.7	Log mean temperature difference LMTD [°C]	30.8

Table 2 compares the optimal designs obtained using the traditional optimization procedure of Caputo et al. (2010), where no uncertainty exist in the value of U when designing the exchanger, and the procedure described in this work, allowing uncertainty in the estimation of the heat transfer coefficient. Different optimal configurations are obtained when the objective function is computed according to different expressions for EP. In particular, here results are shown for the four EP types described above. The Table shows all optimal values of VIP chosen by the GA as well as the main resulting VDP which describe the equipment architecture. Also shown are the capital investment, the discounted operating cost, the economic performance EP and the objective function value.

It should be pointed out that the traditional optimization procedure for an heat exchanger designed with certain value of U is different from the one depicted in Figure 1. In fact, during the design phase the value of EP is zero because all candidate solutions generated by the GA comply with all specifications, and the design is optimized in order to minimize the sole sum of capital investment and the life cycle discounted pumping cost. However, in the evaluation phase that optimal exchanger is subjected to random variation of the actual U value and the expected value of EP is computed so that the actual value of the objective function can be computed. Conversely, when the design optimization procedure under uncertainty is adopted the variation of U is taken into account directly within the design procedure so that the resulting optimal design is obtained by minimizing an objective function which directly includes the EP value.

Table 2. Optimization results.

EP type		I		II		III		IV	
Optimization Type		Traditional	This work	Traditional	This work	Traditional	This work	Traditional	This work
D_b	[mm]	830.00	770.00	760.00	840.00	980.00	1040.00	850.00	890.00
Bcut	[%]	25	25	35	45	35	35	25	15
L_{bc}	[mm]	792.0	836.0	826.0	908.0	694.0	808.0	918.0	959.0
L_{tp} Ratio	[-]	1.25	1.25	1.25	1.33	1.33	1.33	1.25	1.33
d_o	[mm]	38.000	20.000	38.000	12.100	18.000	30.000	14.000	18.000
θ	[deg]	30	90	90	90	30	30	30	90
N_{tp}	[-]	6	4	4	4	6	8	6	4
N_{tt}	[-]	222	637	164	1888	1314	521	1827	952
L_{tt}	[m]	9.77	6.88	15.26	3.74	3.60	6.65	2.84	5.90
h_S	[W/m^2 K]	2971.7	3494.8	2311.2	2592.7	2885.9	2030.3	4322.6	2849.7
h_T	[W/m^2 K]	2358.9	2691.0	2171.8	3107.7	2649.8	2469.9	3730.3	2478.1
U_{dirt}	[W/m^2 K]	693.5	725.4	632.9	690.9	681.5	622.5	789.2	664.3
S	[m^2]	251.88	267.68	291.09	260.67	257.84	317.40	221.30	309.76
DP_S	[kPa]	17.21	16.12	14.64	4.01	7.77	4.91	10.25	6.69
DP_T	[kPa]	29.20	28.92	22.82	33.37	26.94	36.96	57.71	22.83
C_{es}	[€/yr]	14663.6	14108.3	11989.8	9765.5	9908.3	11022.5	18364.0	8443.2
C_I	[€]	70487.3	73826.8	78742.8	72346.4	71748.8	84225.4	63968.9	82637.8
$C_I + C_{es}$	[€]	85150.9	87935.1	90732.6	82112.0	81657.1	95247.9	82333.0	91081.1
EP_{tot}	[€]	45769816.5	46852035.2	-16350.60	0.0	1617599.5	2776075.7	1583011.7	3144272.9
$C_I + C_{es\,+}\,EP_{tot}$	[€]	45684665.7	46764100.1	-107083.2	-82112.0	1535942.40	2680827.8	1500678.7	3053191.8

Results clearly show that the exchanger configuration obtained considering the uncertainty of U during the design phase always has a superior economic performances respect an exchanger optimized assuming a certain value of U but operated with uncertain heat transfer coefficient value. This confirms the validity of the proposed methodology.

4. CONCLUSIONS

Considerable uncertainty affects the computation of heat transfer coefficients and the design of heat exchangers. This determines a conservative design approach which can be penalizing under the economic point of view. In this work the uncertainty in heat transfer calculations is included in the design procedure and a design optimization method is utilized which tries to optimize the economic performance of the equipment assuming that it operates in off design conditions owing to the uncertainty in the actual value of the heat transfer coefficient. This allows to design a robust heat exchanger with superior performances respect exchangers optimized assuming no uncertainty in design condition.
To support the proposed approach a case study demonstrated that exchangers sized for uncertain heat transfer conditions have superior economic results (higher revenues or lower life cycle costs) respect the corresponding exchangers sized referring to nominal heat transfer conditions but operated under uncertain conditions.

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